NSWCCD-83-TM-2013/35 Initial Investigation of Wave Impact Load Transfer Through Shock Mitigation Seats in High-Speed Planing Craft

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INITIAL INVESTIGATION OF WAVE IMPACT LOAD TRANSFER THROUGH SHOCK MITIGATION SEATS IN HIGH-SPEED PLANING CRAFT

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This report summarizes initial investigations of wave impact load transfer through passive shock mitigation seats used in high-speed planing craft. Data recorded during seakeeping trials are used to illustrate methods for developing transfer functions that characterize shock mitigation seats, and the dynamic response of compliant seat cushions is explained using data from multiple sources. This is part of a broader study of the transfer of wave impact load from the keel up through the craft into various systems.

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Introduction

Objective

This report summarizes initial investigations of wave impact load transfer through passive shock mitigation seats used in high-speed planing craft. Data recorded during seakeeping trials are used to illustrate methods for developing transfer functions that characterize shock mitigation seats, and the dynamic response of compliant seat cushions is explained using data from multiple sources. As part of a broader study investigating the transfer of wave impact load from the keel up through the craft into various systems of interest, these results should be useful not only for evaluating the efficacy of shock mitigation seats, but also for the study of long term adverse health effects on seat occupants in severe wave slam environments.

Scope

This report focuses only on passive shock mitigation seats consisting of mechanical systems with springs and dampers (i.e., manually adjustable or not adjustable, and with or without foot rests). Semi-active seats, active seats, and jockey seats are not addressed. Topics related to biomechanics and adverse effects on human health due to one severe wave impact in a seat or repeated impact exposure over time are not specifically addressed, but the use of wave impact acceleration data in human health assessment is summarized as background information.

The wave impacts described herein are based on studies of acceleration data recorded during seakeeping trials of manned and unmanned high-speed planing craft in rough seas. The database includes twenty-one craft that weighed approximately 14,000 pounds to 116,000 pounds and had lengths that varied from 33 feet to 82 feet.

Background

The International Organization for Standards (ISO) published in 2004 revision “E” of ISO Standard 2631-5. It provides guidance for estimating adverse health effects on the lumbar spine for a seated person as a result of exposure to whole body vibrations that contain multiple shocks [1]. Compression of the spine is of primary interest for exposure severity. The method uses the peak acceleration response of the lumbar spine to compute a spinal dose over time.

If accelerations are measured at the lumbar spine location the data can be fed directly into an algorithm that computes an equivalent acceleration dose for an estimated eight hour period. This type of data can be obtained through the use of a tight fitting belt, commonly referred to as a kidney belt, with imbedded tri-axial accelerometers. Another option is to acquire data during tests using an anthropomorphic test device (ATD) that has imbedded accelerometers or load cells. These are the most direct approaches to obtaining accelerations or forces at the lumbar position.

If kidney belt or ATD data is not available, or if there is no other means of recording peak lumbar spine responses, the remaining alternative is to record the data at another location. For example, acceleration data could be recorded on top of a seat cushion beneath the seat occupant. The cushion data (i.e., seat pad data) would then be used as input into a mathematical model that would estimate the lumbar spine response, which in turn would be used in the algorithm that estimates spinal dose.

In ISO 2631-5 the assumed measurement location is on top of a seat cushion. Peak accelerations measured on the seat cushion are then used as inputs into a mathematical model that estimates the lumbar spine peak acceleration response. Cushion peak accelerations are the input, and lumbar peak accelerations are the output. In classical signal processing theory this input – output relationship is the basis for defining a transfer function between the input and output signals.

The following paragraphs provide useful information and recommendations for processing and interpreting acceleration data signals as they pertain to wave impact load transfer, transfer functions, and seat cushion dynamics. The underlying focus is on the transmission of impact forces during wave impacts from the keel of a craft, up through the structure, and into systems of interest, including structure, equipment, shock mitigation seats, and personnel effects.

Transfer Functions

Frequency Domain Signal Processing

The term "transfer function" is used in the analysis of systems such as single-input single-output filters in the fields of signal processing, communication theory, and control theory [2]. For discrete-time systems that are linear, time invariant systems, the transfer function is the ratio of the Laplace transform of the output signal divided by the Laplace transform of the input signal. Similar transfer functions can also be developed using Fourier transforms of the input and output signals. The transfer function is characterized in the frequency domain by a plot of amplitude versus frequency and phase-angle versus frequency. Equation (1) shows the mathematical form of the transfer function H(s) computed from a ratio of Laplace transforms.

$$H(s) = \frac{Y(s)}{X(s)} = \frac{\Im(y(t))}{\Im(x(t))} \qquad \text{Equation (1)}$$

For high-speed craft applications, in theory, the ratio of the Laplace transform of the lumbar acceleration (i.e., output) and the Laplace transform of the seat cushion acceleration (i.e., input) will provide an amplitude versus frequency plot and a phase angle versus frequency plot. These two plots constitute a "transfer function" that can be used to estimate the lumbar acceleration for a seat cushion acceleration input. The characteristics of the frequency plots can also be described as infinite impulse filters for certain linear, time invariant systems.

In theory, the concept is like filtering an input signal. The filter transfers the input signal to the output signal. Thus, the input seat cushion acceleration can be passed through a filter to yield an estimated lumbar acceleration response. Equation (2) shows that this requires an inverse

Laplace transform of the product of the transfer function and the Laplace transform of the input acceleration.

$$y(t) = \Im^{-1}\{H(s)\,\Im(x(t))\} \qquad \text{Equation (2)}$$

The transfer function H(s) is a way of assessing the effectiveness of shock mitigation seats [3]. In practice, the ability to use Laplace transforms or Fourier transforms and inverse-Laplace transforms and inverse-Fourier transforms to develop transfer functions using craft acceleration signals or land-based test signals is unknown. It is not clear whether seat cushion data recorded under different speed and wave height conditions on different seats in a craft or in a laboratory would provide the same transfer function. The mathematical equations are complicated, and it is not clear that inverse-Laplace transforms or inverse-Fourier transforms provide mathematically unique solutions for time variant signals.

The original transfer function in the 2004 ISO standard is referred to as a recurrent neural net that has characteristics of a non-linear finite impulse response filter. Since it was developed for ground vehicle applications using data with peak accelerations less than 4 g (with no specification of impact duration), another approach for predicting larger amplitude lumbar spine responses was recommended for high-speed craft applications [4]. The approach avoided complications in the frequency domain by simply dividing peak lumbar accelerations by peak seat cushion accelerations in the time domain.

Time-Domain Lumbar Response

The time-domain approach recommended in 2008 is shown graphically in Figure 1. The figure is a plot of lumbar (L4) peak acceleration versus seat-cushion peak acceleration. It includes data recorded on a shock mitigation seat during seakeeping trials of a high-speed craft as well as peak lumbar accelerations predicted by a mathematical model of the lumbar spine (using MADYMO™ software). It is assumed that the MK V lumbar data was obtained using kidney belt accelerometers.

Two straight lines are shown in the figure based on MADYMO™ calculations. Both are within a few percent of the y = x line. In other words, the response at the lumbar spine is about the same as the response at the seat cushion. For this data set the “transfer function” is approximately 1.0.

The essential importance of Figure 1 is the process used to develop the transfer function. In this x – y format, each data point is the peak acceleration at location “y” plotted on the x-axis at the corresponding peak acceleration at location “x”. The least-squares data fit through all data points provides an equation to estimate peak acceleration at location “y” if the peak acceleration at location “x” is known. This empirical approach to assessing seat performance is at least as relevant as a complex transfer function in the frequency domain [3]. Caution is advised however when processing data in the time domain and using the term “transfer function”. Since “transfer function” is taught in many engineering and mathematics courses as a frequency domain process, it may be more descriptive in the time domain to use the terms scale factor or response ratio to describe the response relationships between two different locations (e.g., the seat cushion response and the lumbar acceleration response).

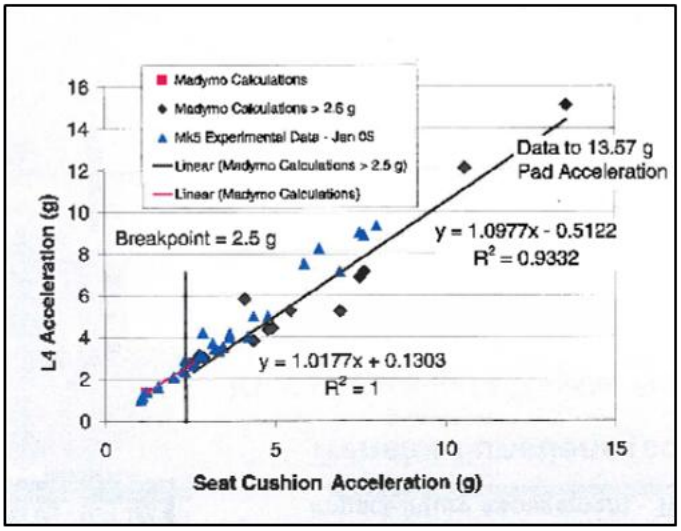

Figure 1. Lumbar L4 Acceleration versus Seat Cushion Acceleration

Example Time Domain Application

In the absence of kidney belt data, the data fitting approach shown in Figure 1 will be illustrated in an example using acceleration data recorded at two different locations for a passive shock mitigation seat. The unfiltered acceleration data is shown in Figure 2. It was recorded during seakeeping trials of a planing craft in short wave-period head seas while a person occupied the seat. One gage was located on the deck of the craft at the base of a shock mitigation seat; the other gage was attached under the seat pan. Both gages were oriented in the vertical direction. The insert in the figure shows an expanded time-scale of acceleration responses during and between three successive wave impacts. The three large red spikes were caused by the seat pan impacting the bottom of the seat structure.

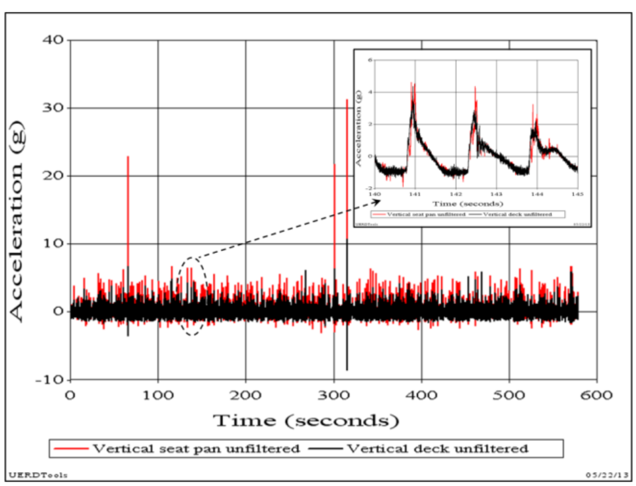

Figure 2. Deck and Seat Pan Unfiltered Vertical Acceleration

Frequency Domain Analysis

Even though the example is presented in the time-domain in terms of peak accelerations, it is important to understand the frequency content of the unfiltered acceleration signal. Knowledge of the frequency content in the acceleration signal enables proper application of the data to the intended use.

Close inspection of both acceleration records shows small amplitude oscillations that are often referred to as signal “hash” or “noise”. The physical sources of these oscillations are the vibrations of the structure at each gage location. Figure 3 shows the Fourier spectrum of both acceleration records. The insert in Figure 3 is an expanded frequency scale of the Fourier spectrum to better show the vibration frequencies. On the seat pan (i.e., the red curve), the pan structural vibrations appear to be roughly in the 37 Hz to 45 Hz range. The hump in the vicinity of 12 Hz to 14 Hz is caused by the relative motions of the spring-damper assembly between the deck and the seat pan. In the deck acceleration signal the vibration content is seen roughly in the 28 Hz to 30 Hz range. The slight hump at 14 Hz in the deck spectrum is believed to be due to deck motions caused by the oscillation of the seat assembly. In other words, 12 Hz to 14 Hz motions of the seat assembly fed back into the deck structure as in a two degree-of-freedom system.

Modal decomposition of the unfiltered acceleration records was performed to separate the original signal into its fundamental components, including the rigid body component, relative spring oscillations, and structural vibrations. The rigid body deck acceleration during each wave impact period is directly related to the force of the impact, so it is a measure of the impact load in units of “g” [5].

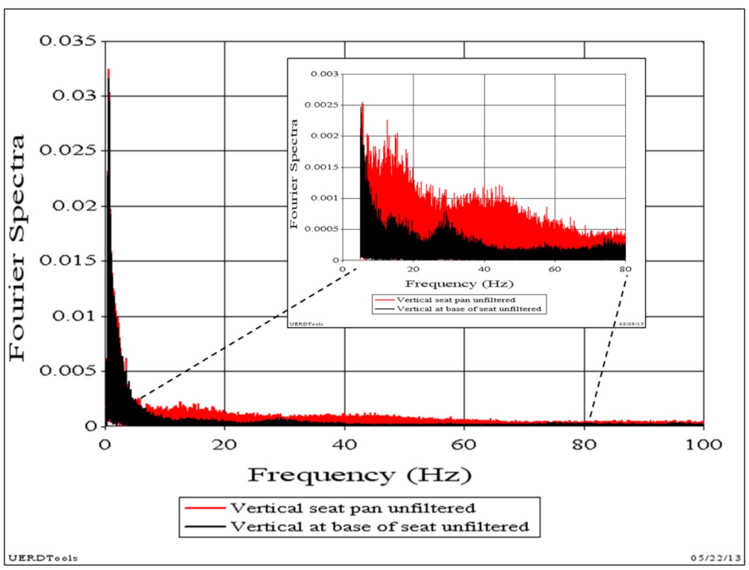

Figure 3. Fourier Spectrum of Deck and Seat Pan Acceleration Signals

Peak accelerations in the vibration component of the signal are the rate of change of the velocity of very small oscillations in the deck that do not transfer momentum from the deck to the seat assembly or from the pan to the occupant. The oscillations are on the order of less than 0.05 inches for frequencies greater than 30 Hz. The best mathematical model of input acceleration and response acceleration of an impulsive load should be based on the relevant modes. The rigid body mode is the dominant relevant mode for the deck input acceleration, and the rigid body mode plus the relative displacement mode are the dominant relevant modes for the seat pan data. In other words, the vibration content of the acceleration records should be filtered out before peak accelerations are tabulated and plotted to estimate ratios between input and output locations.

Peak Acceleration Comparisons

In the following data plots, the peak deck accelerations are based on use of a 10 Hz low-pass filter to capture only the dominant rigid body impacts observed in Figure 3 at less than 2 Hz. The seat pan acceleration is low-pass filtered at 30 Hz to remove the vibration content and to preserve the rigid body content and the spring-damper relative motion components. Figure 4 shows the low-pass filtered accelerations for the three wave impacts shown in the insert in Figure 2.

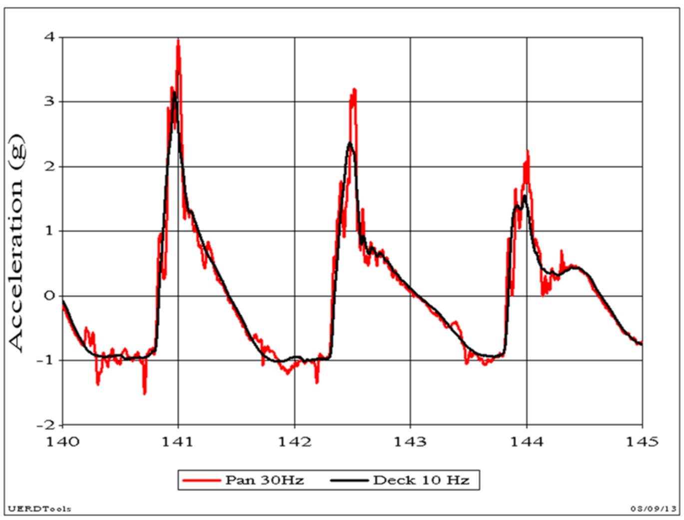

Figure 4. Low-Pass Filtered Acceleration Records

Figure 5 shows the filtered peak pan acceleration compared to the input peak deck acceleration for each wave impact. Only data recorded between time 100 msec to 400 msec were processed to minimize the time to construct the plot. The red “X” data points correspond to wave impacts that caused seat bottom impacts. Bottom impacts did not occur for the other (blue circle) data points.

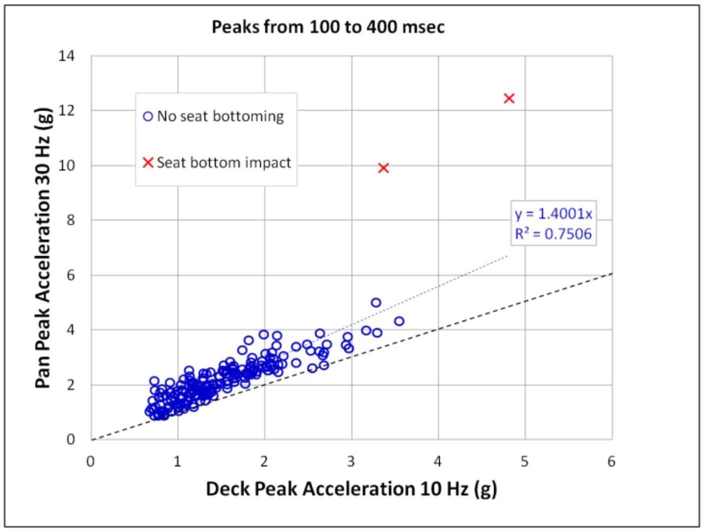

Figure 5. Peak Pan Acceleration versus Peak Deck Acceleration

The least squares data fit line with a zero intercept for the “no seat bottom” events has a slope of 1.40. In other words, on average, the peak pan accelerations were forty-percent higher than the corresponding peak deck input acceleration. If this craft were tested again in different sea conditions, it is assumed that the equation y = 1.4x could be used to estimate peak pan accelerations (y) for each peak deck acceleration (x). In this example, the data shows that the deck peak accelerations are amplified by the seat spring-damper assembly, even during non-bottom impact events, so shock mitigation was not achieved.

Displacement Amplitudes

The frequency domain of interest in ISO 2631-5 2004 (E) is 80 Hz and below [1]. It is assumed that this is based on the original transition of the study of whole-body vibration effects on humans to the study of adverse health effects caused by “vibrations with multiple impacts”. In the frequency domain, it is understood that a broad-band analysis is appropriate to adequately characterize complex transfer functions in the frequency domain (i.e., based on ratios of Laplace transforms and inverse Laplace transforms or Fourier transforms).

In the time domain, it is important to relate the physics of dynamic motions to impulse and momentum relationships. Frequencies are important, but the potential for human discomfort or adverse health effects are also a strong function of relative displacements. For example, if pure sine waves with frequencies of 60 Hz and 80 Hz and peak acceleration amplitudes of 10 g are of interest, Table 1 shows that theses oscillations have relative displacements of +/- 0.026 inches and 0.015 inches, respectively. Likewise, if 4 g vibrations are of interest over a range from 20 Hz to 80 Hz, these oscillations have displacements of +/- 0.097 inches to 0.006 inches. These are extremely small compared to craft heave displacements and relative displacements of shock mitigation seats.

Table 1. Frequency and Displacement Values

Frequency Hz	Acceleration (g)	Velocity (fps)	Displacement (inches)
20	4	1.02	0.097
30	4	0.67	0.041
60	4	0.34	0.010
80	4	0.25	0.006
20	10	2.5	0.240
30	10	1.7	0.100
60	10	0.85	0.026
80	10	0.64	0.015

Modal Decomposition

Figure 6 shows plots that illustrate modal decomposition of the unfiltered acceleration records into three distinct modes of response. The three modes added together yield the original unfiltered acceleration records. The rigid body mode has wave encounter frequencies of roughly 2 Hz or less. For this portion of the acceleration signal the craft's heave displacements in the time history domain can be on the order of 6 inches to 4 feet (i.e., as in a free-fall drop heights) just prior to impact, and the craft can plunge into the water during the impact for roughly 6-inches to 18-to-22 inches depending upon craft weight and speed. These displacements characterize the energy and momentum transfer that occurs during a wave impact.

When subsequent forced vibrations become part of the acceleration time history response, the effects of the impulsive load must be analyzed with an understanding of the vibration displacements. Figure 6 shows spring-damper oscillations of the seat pan on the order of +/- 1 g and less with response frequencies in the 12 Hz to 14 Hz range. These relative displacement motions would be on the order of 0.07 inches and less. The seat pan vibrations are on the order of +/- 1.5 g with frequencies on the order of 37 Hz to 45 Hz. These displacements would be on the order of 0.01 inches and less. It is not intuitive that 0.01 inch to 0.07 inch contributions to combined heave displacements and spring-damper relative displacements could play a key role in human comfort or the potential for adverse health effects, especially when cushions are employed to avoid hard seat environments.

Unfiltered Data and 80 Hz Filtered Data

Figure 7 compares the unfiltered pan peak accelerations for the acceleration record shown in Figure 2 with values obtained using 80 Hz and 30 Hz low-pass filters. The vibration content can be seen as the vertical differences between the plots. Each curve is the plot of the peak accelerations sorted largest-to-smallest from left to right.

Figure 8 shows the same data from Figure 7 plotted in a different format. The largest pan acceleration and deck acceleration are plotted as an x-y pair, then the second largest are plotted as an x-y pair, and so on. It shows that for peak pan accelerations greater than 2 g, the unfiltered peaks are on average at least forty-five percent greater than the 30 Hz low-pass filtered values (i.e., y = 1.45x). The 80 Hz low-pass filtered peaks are about twenty-five percent greater (i.e., y = 1.25x).

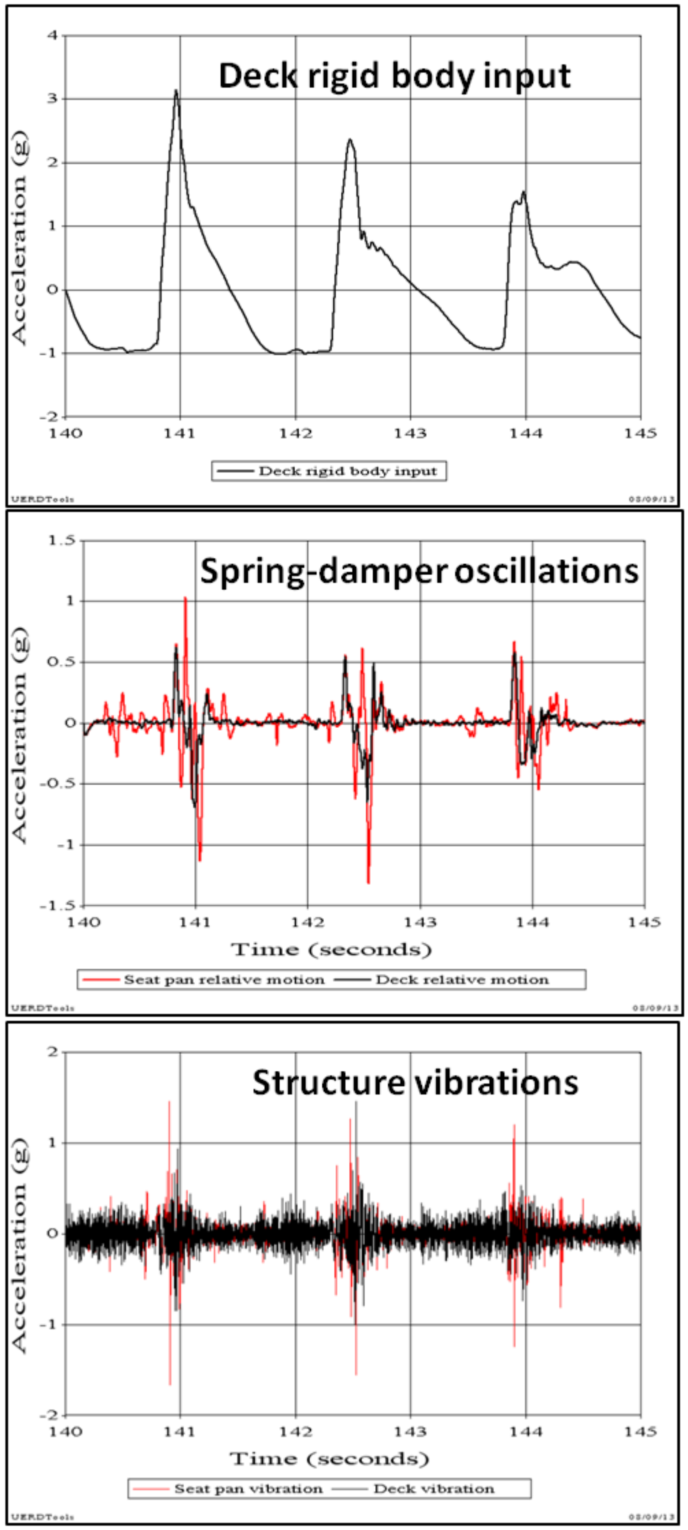

Figure 6. Modal Decomposition of Acceleration Signal

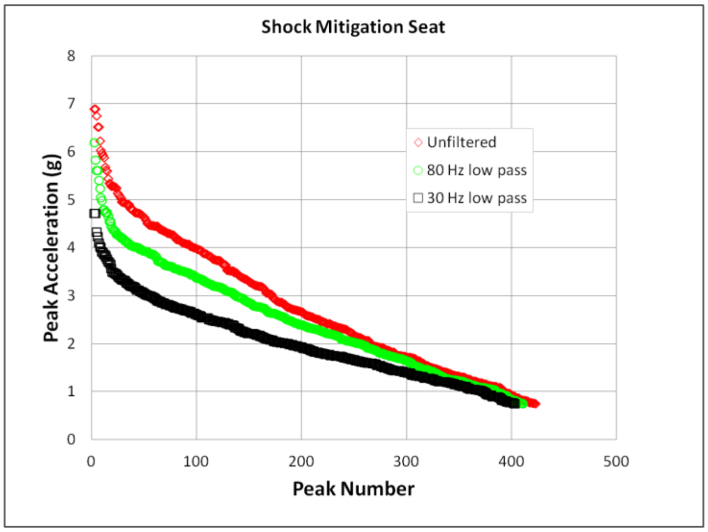

Figure 7. Low-Pass Filtered and Unfiltered Peak Pan Accelerations

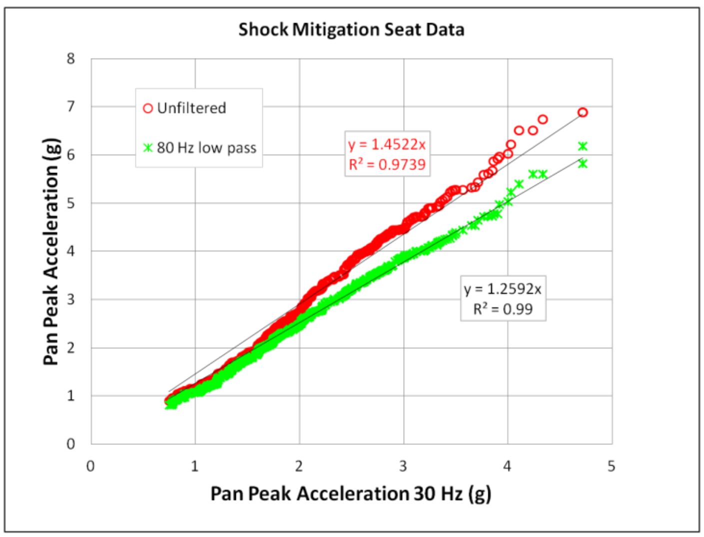

Figure 8. Unfiltered and 80 Hz Peak Pan Accelerations

Data Plotting

Figure 5 was created by plotting peak pan and peak deck accelerations for individual wave impacts. This can be a time consuming manual process without an algorithm specifically created for pairing-up peak accelerations from two different acceleration time signals that are synchronized in time (i.e., each data point is an x-y pair for one wave impact). This limitation can be obviated by using a different plotting approach.

The alternative approach to estimating the deck-to-pan scale factor is to use the Ride Severity Index (RSI) approach [6]. The RSI is defined as the slope of the linear least-squares data fit to peak accelerations with an intercept of zero. In this approach the peak accelerations for the deck greater than the root-mean-square (RMS) acceleration are sorted and listed largest to smallest. Likewise, the list of largest-to-smallest peak pan accelerations is then created. The largest peak pan acceleration is plotted with the largest deck acceleration as an x-y pair. Then the second largest pan acceleration is plotted with the second largest deck acceleration, and so on, until all x-y pairs are plotted. In this approach the peaks are not time-synchronized. The result is an average of the ride severity over a period of time. Figure 9 shows how the RSI x-y pair data points (shown in red) compare to the time-synchronized data (shown in blue). This approach is merely a convenience that lends itself to rapid data processing with spreadsheet applications.

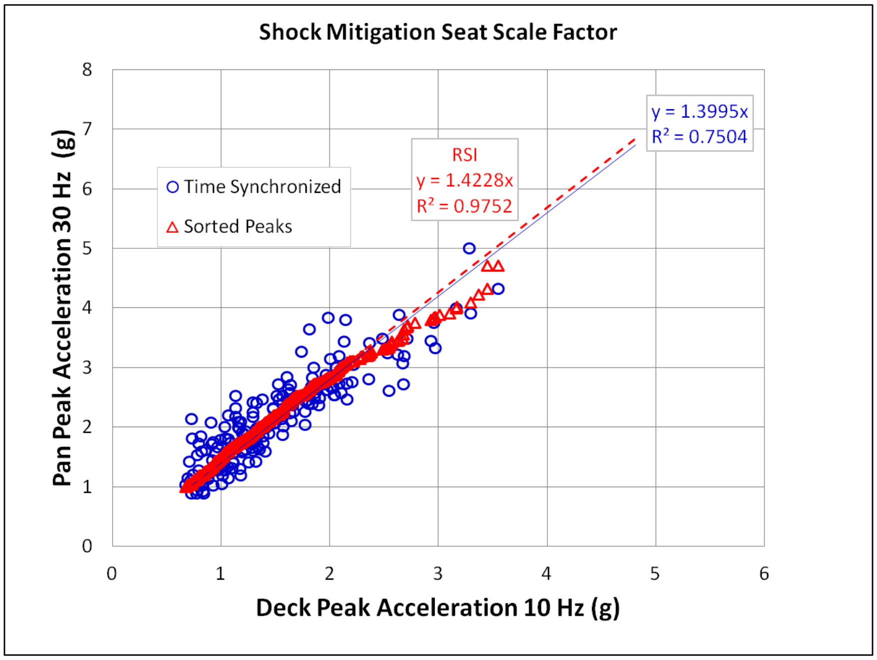

Figure 9. Ride Severity Index (RSI) Averaging Technique

In this example, the Ride Severity Index between the deck and the pan is 1.422, compared to the time-synchronized ratio of 1.399. The higher severity impacts greater than 3 g have a ratio closer to 1.25, so the linear plotting approaches may yield a conservative ratio (i.e., scaling function) for some data sets. If large variations from the linear trend occur in the data, a piece-wise or segmented linear data fit can be developed [4].

Limitations

The deck and seat pan data shown in Figure 2 provide data for scaling from the deck to the seat pan for specific conditions. The word "scaling" is used here as a synonym for the word "ratio", as in y/x = 1.40 for the ratio of the 30 Hz pan peak accelerations (y) to the 10 Hz deck peak accelerations (x). This ratio is applicable only for seat installations and test conditions under which the data was obtained, either in a laboratory or during seakeeping trials. The following limitations are presented merely to illustrate the approach to developing scale factor limitations for other data acquisition scenarios.

Seat Manufacturer

Passive shock mitigation seats come in a variety of configurations with different spring-damper assemblies, different support structure, and different pan structural configurations. The empirical data ratio (i.e., scale factor) is applicable only to similar installations with equivalent spring-dampers (i.e., similar stiffness and damping coefficients) and similar frame and seat masses.

Wave Impact Duration

Wave impact severity is quantified by both the peak acceleration and duration of the impulse. Figure 5 shows that the example deck-to-pan ratio is relatively constant over a range of peak deck accelerations from 0.75 g to 3 g (10 Hz low-pass filtered). This defines the range of applicability of the ratio as long as wave impact durations do not vary significantly (assumed to be within 0.10 seconds to 0.40 seconds for 25 foot to 50 foot craft that weigh less than roughly 50,000 pounds). Spring-damper characteristics vary significantly as impulse duration decreases below 0.10 seconds, especially as durations approach 0.05 seconds or lower, so scale factors should not be developed for impulses less than 0.1 seconds for high-speed craft that weigh less than approximately 50,000 pounds (unless so indicated by data).

Seat Bottom Impacts

Figure 5 shows that impacts above the 3 g level can lead to seat bottom impacts, so the scale factor should not be used to extrapolate above the level where bottom impacts are observed.

Seat Cushion – Lumbar Characteristics

If an empirical scaling function is based on the ratio of measured seat cushion peak accelerations and measured lumbar peak accelerations, the applicability of its use for multiple types of seats and multiple seat occupant characteristics should be bounded, preferably based on empirical results.

Seat Cushion Dynamics

Shock Mitigation Seat Data

Figure 10 shows vertical acceleration responses recorded during rough-water seakeeping trials of a 45-foot high-speed planing craft. The three accelerometers were installed on the deck at the base of a shock mitigation seat, underneath the metal seat pan, and inside a rubber pad positioned on top of the seat cushion directly under the seat occupant. The data was subjected to a 30 Hz low-pass filter to remove high-frequency vibrations in order to more clearly see the relevant deck, pan, and pad accelerations. The interesting observation is that the peak acceleration measured on the seat pad (i.e., above the cushion) is larger than the peak acceleration measured on the deck and on the seat pan.

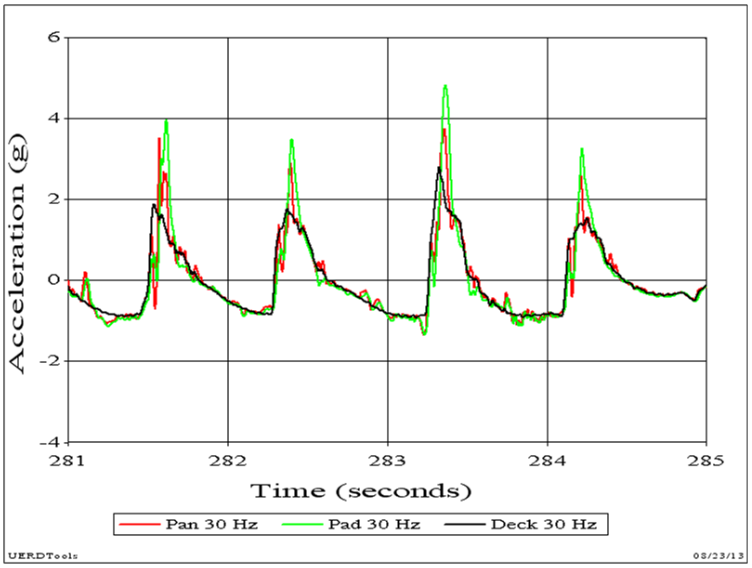

Figure 10. Shock Mitigation Seat Vertical Acceleration Data

In this data set, peak accelerations on the seat pan are larger than peak accelerations on the deck due to dynamic amplification. Dynamic amplification occurs when a structure's natural response period (i.e., inverse of system natural frequency) is less than the duration of the impulsive load [7]. It is not as immediately obvious why the pad accelerations are larger than seat pan accelerations, especially when a common perception would be that a soft seat cushion should reduce accelerations rather than amplify them. But the perception does not take into account the duration of the dynamic load or relevant velocities and displacements. The dichotomy between the common perception and the recorded high-speed craft data is best illustrated by cushion data recorded during simulated airplane crash tests sponsored by the Federal Aviation Administration.

FAA Crash Test Results

The Federal Aviation Administration (FAA) of the U.S. Department of Transportation established a testing protocol to evaluate airplane seats and restraints. The purpose of the testing is to demonstrate structural strength and the ability of a seat to protect an occupant from spine and head injuries in a crash environment [8]. Figure 11 shows the test setup for one of the FAA tests known in the airline community as the 60-degree pitch test. The criterion for acceptance in this test set-up is lumbar spine load recorded within the instrumented anthropomorphic test device (ATD). The seat back is angled 13-degrees with respect to the horizontal, so the vector of the horizontal impact force is 17-degrees off the simulated vertical axis (i.e., 30 degrees minus 13 degrees). The test set-up employs a sled device that accelerates to a constant 35 fps speed before impacting a device that simulates the impulsive load shape and amplitude of a 60-degree airplane crash. The properties of seat cushions were known to have a strong influence on the lumbar load performance [9 - 14] so recent calibration testing of the new FAA protocol performed by the FAA's National Institute for Aviation Research included both hard-seat (shown on the right in Figure 11) and cushioned-seat configurations (shown on the left) [15]. A soft 4.5-

inch non-flotation foam cushion (i.e., unlike those used in the airline industry) was used in the test. Figures 12 and 13 show examples of data recorded during the simulated crash tests.

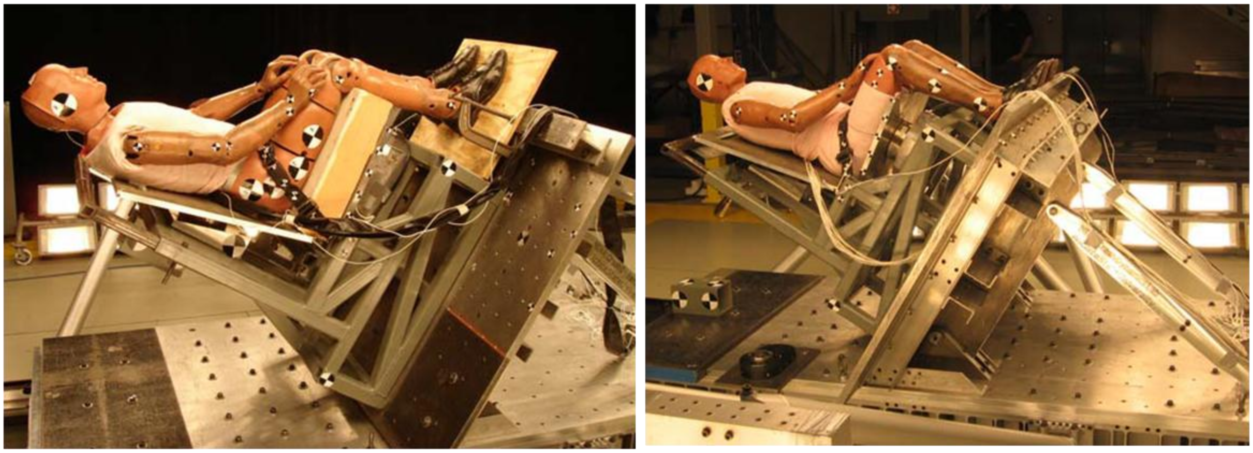

Figure 11. Test Set-up for Cushioned Seat and Hard Seat Configurations

One of the acceleration pulses prescribed by the FAA for seat testing is the isosceles triangular pulse shown in Figure 12. It has a peak acceleration of at least 14 g's that is reached not more than 0.08 seconds after impact. The area under the curve must correspond to not less than 31 fps [16]. The two curves show the sled's good repeatability in generating the acceleration crash pulse. The total duration of the acceleration pulse is approximately 0.16 seconds. This falls within the 0.10 second to 0.40 second nominal range typically observed for wave impact acceleration pulses for high-speed planing craft [7], so the test results are of interest for high-speed craft applications.

Figure 13 shows dynamic forces recorded by load cells during the sled test. One cell was positioned directly under the seat pan to measure the seat pan force perpendicular to the seat, and the other was located within the anthropomorphic test device (ATD) as a measure of axial spine force. The black curve (i.e., with triangle symbols) shows forces for the hard seat configuration, and the red curve shows the cushioned seat results. Both curves were digitized from the original report.

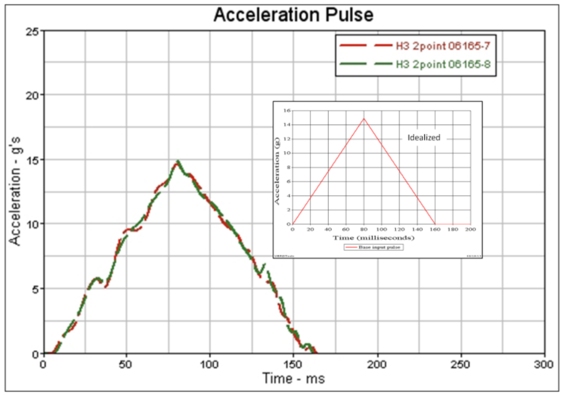

Figure 12. Sled Acceleration Pulse for Simulated Airplane Crash Test

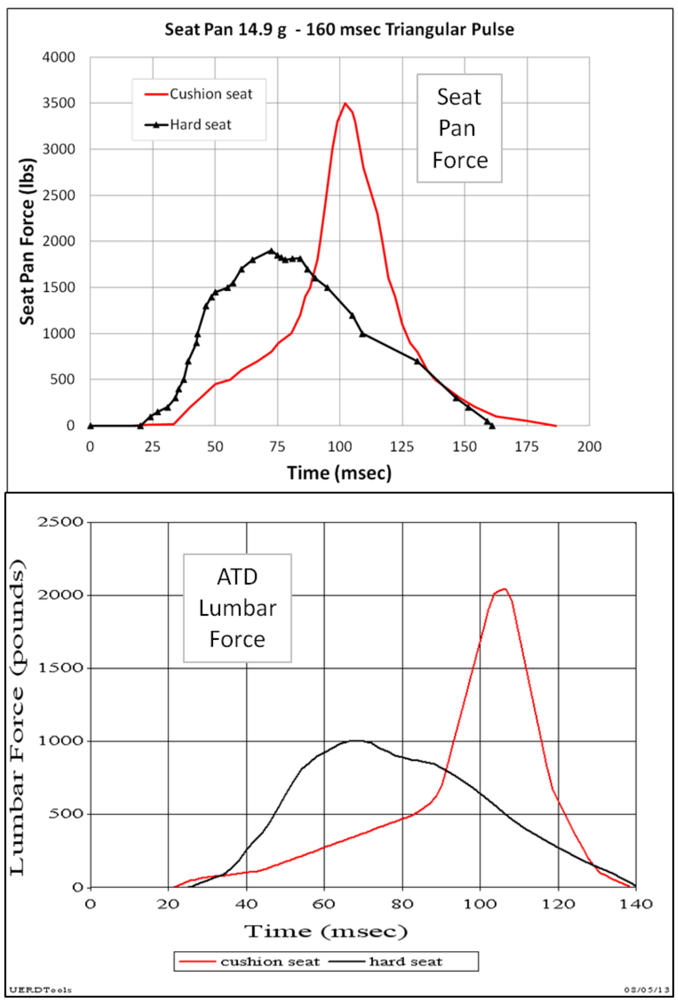

Figure 13. Cushioned Seat and Hard Seat Crash Test Responses

Figure 14 shows the same data from Figure 13 plotted to compare seat pan forces with ATD lumbar forces.

The measured forces for both the seat pan position and the ATD lumbar spine location clearly show that larger forces occur during the seat cushion test compared to the hard seat test. The explanation for the higher seat cushion forces has to do with the compliance (i.e., inverse of stiffness) of the seat cushion. The ATD and the seat/sled assembly reach a velocity of 31 fps prior to the crash impact. When the impact occurs, the seat pan and sled assembly immediately begin to decelerate (i.e., positive force in Figure 13), but as the cushion compresses, the force on the seat pan is smaller than the hard seat configuration, thus the momentum transfer is delayed until the cushion is more fully compressed. The time and shape of the delayed compression depends upon the force-deflection characteristics of the seat cushion material [17]. The momentum of the simulated seat occupant (i.e., the ATD) in the hard seat is the same prior to impact as the cushioned seat test. But, in the cushioned seat test the majority of the momentum transfer occurs over a shorter period of time. As shown in Figure 15, the calculated impulses delivered to the seat occupant are the same for both hard and cushioned seat tests, but the seat occupant in the cushioned seat receives the majority of the change in impulse over a shorter period of time, thus the force is higher (i.e., higher impulse slope) over the shorter period of time.

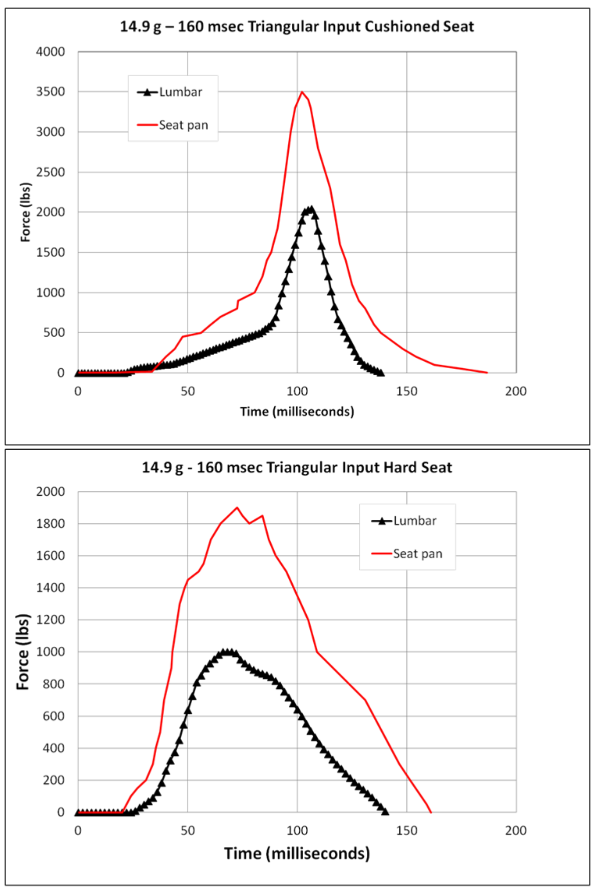

Figure 14. Seat Pan Force vs. ATD Lumbar Force

The test results confirm that seat cushion selection is a balance between occupant comfort during non-impact conditions and occupant safety during impacts. Although the softer cushion solution provides comfort for prolonged periods, the softer cushion is not recommended because the low stiffness usually makes them more hazardous during impact conditions [12].

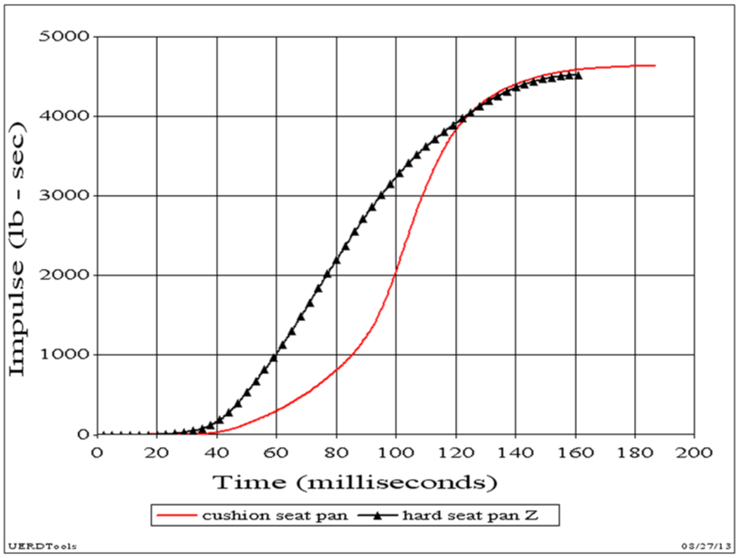

Figure 15. Seat Pan Impulse with and without Cushion

The measured load cell data presented in Figures 13 and 14 presents a clear relationship between forces acting above and below the cushion. These relationships are very useful for understanding acceleration data that is recorded above and below a seat cushion in a shock mitigation seat installed in a high-speed planing craft.

Craft Seat Cushion Dynamics

Figure 16 shows the deck, seat pan, and pad acceleration responses for the wave impact observed in Figure 10 between 283 seconds and 284 seconds, as well as the integrated velocity responses. The unfiltered acceleration records were subjected to a 30 Hz low-pass filter to remove deck and pan vibration accelerations. The peak acceleration measured on the deck is 2.8 g. On the seat pan it is 3.7 g, and on the seat pad it is 4.8g. The acceleration responses on the seat pan and the pad show the same phenomenon as the FAA force data in Figure 13. The compliance of the spring-damper assembly and the compliance of the seat cushion results in a delay in the transfer of the pre-impact momentum of the seat occupant. This can also be seen in the velocity time history as the longer delay in time and therefore a steeper slope for the negative velocities for the seat pan and pad to change to zero velocity. When the deck reaches zero velocity, an increased compression in the seat spring-damper assembly and in the cushion is required to rapidly decelerate the pan and the pad to zero velocity. Thus, instead of the force of the impact being distributed evenly over time, more force is applied to the seat occupant closer to the end of the impact. Since the responses must have the same change in impulse (i.e., area under the impact acceleration curves), a larger force must be applied to the occupant closer to the end of the impact than if no cushion had been used.

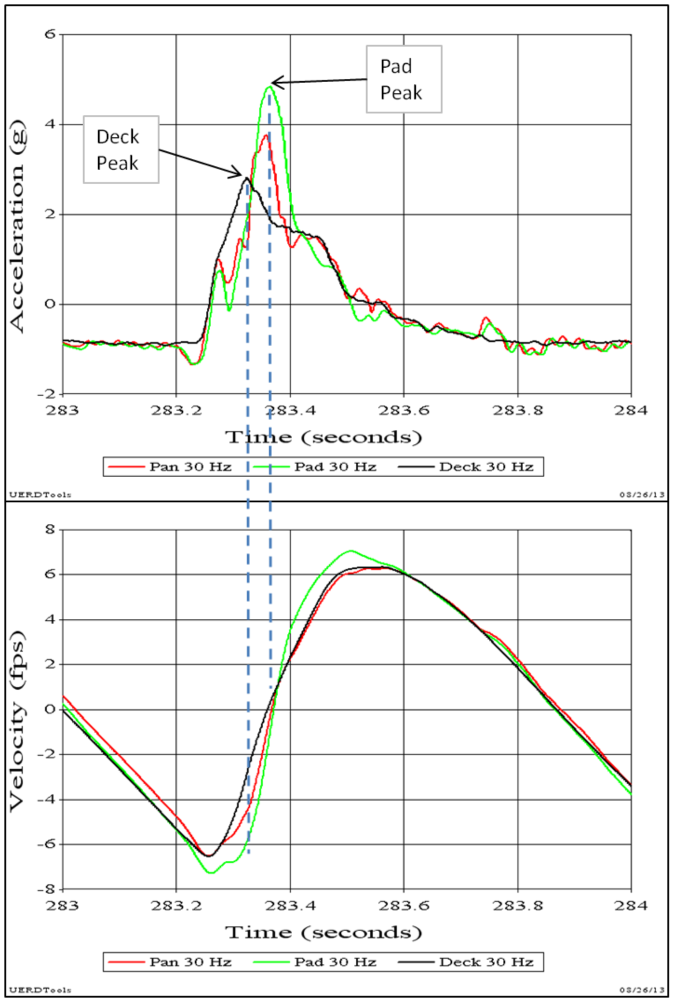

Figure 16. Slam 283 Deck, Pan and Pad Acceleration Responses

In Figure 16 the data shows that one additional subtle force is acting besides the force of the water impact and the delay in the momentum transfer. At the time of water impact the deck acceleration is approximately minus 0.9 g (i.e., almost a pure free-fall event). At the same time the acceleration of the seat pan and the pad is minus 1.3 g. The larger negative acceleration of the seat pan and pad is assumed to be caused by the increased downward force when the spring is fully extended and the upward velocity of the seat occupant is zero (when the lap belt is fully stretched in tension and the strain energy stored in the spring is released). This effect can be exaggerated when the rate of downward bow pitching is large prior to impact. The force of the extended spring increases the downward acceleration of the seat pan to be larger prior to impact. The impulse delivered to the seat occupant is therefore slightly higher than the impulse delivered to the deck by the wave impact.

The blue dotted line to the right in Figure 16 is positioned at the time when peak accelerations (and therefore inertial forces) on the pan and the pad are largest. Since the deck has already started moving upward at this point in time, and the spring and cushion are fully compressed, the largest upward force through the pan and the pad is observed as the largest peak acceleration.

Seat Cushion Lessons Learned

The seat pad accelerations shown in Figure 10 and Figure 16 suggest that the same lessons learned in the aviation industry for airplane ejection and crash impacts also apply for wave impacts in high-speed planing craft. These lessons are summarized below.

Seat cushions are primarily designed for comfort. Their form fitting characteristic spreads the occupant load over the largest possible area in non-impact environments thereby decreasing high pressure points and preventing restriction of blood flow [12].

Every effort should be made to design a cushion that minimizes relative motion between the occupant and the seat and that acts as a shock damper between the occupant and the mass of the seat [10]. Otherwise impact force (or acceleration) amplification can occur [9, 10, 11, 12, 15, and 16].

Relative motions can be minimized by increased foam density and/or reduced foam thickness [17].

Different layers of viscoelastic and loading-rate-sensitive materials can be used to achieve these goals [12, 13].

Dynamic analyses and/or testing should be conducted to demonstrate that the cushion design produces a compromise between crash (i.e., impact) safety and comfort over the operational and dynamic conditions of interest [12, 17].

Cushion comfort is of primary concern and must not be unduly compromised to achieve crash (i.e., impact) safety [12, 14, and 18].

Observations and Recommendations

The purpose of a shock mitigation seat in high-speed planing craft is primarily to limit the vertical load experienced by the seat occupant. The most direct approach to monitoring transmitted load is through the use of load cells in the seat assembly or in anthropomorphic test devices (ATD). In the absence of load cells, vertical accelerometers can be positioned to measure the transfer of inertial loads from the deck, to the seat pan, and into cushion pads or kidney belts worn by human occupants or ATDs.

The ISO-2631 Part 5 algorithm for estimating the potential for adverse effects on the spine due to impact exposure is based on a peak acceleration dose at the lumbar spine position. The most direct approach to measuring lumbar peak acceleration is through the use of an accelerometer positioned at or near the lumbar position. This obviates the need for a mathematical transfer function for scaling peak accelerations from another location to the lumbar position.

If data scaling from the seat pan or the seat cushion to the lumbar is required, the most practical approach is by developing empirical scaling functions as shown in Figures 1, 5, or 9. This approach avoids the complications of complex Laplace or Fourier transfer functions in the frequency domain.

Prior to creating mathematical scaling functions, acceleration data should be decomposed into its fundamental modes to understand the signal's content. In the context of impulsive shock loads, acceleration signal content due to structural vibrations is considered noise that is unrelated to the transfer of the impulsive load (i.e., transient wave impact load) that can affect occupants in shock mitigation seats. These very small transient vibrations of the deck or seat pan in the vicinity of the gage (e.g., less than 1/32-inch) have little to no effect on the response of a seat occupant to the impulsive load, especially in the presence of seat cushion materials. They therefore have little relevance in the study of adverse effects on the spine due to wave slam effects. Modal decomposition of acceleration data should be used to remove vibration acceleration content in order to properly quantify the amplitude of measured inertial accelerations (i.e., loads in units of "g"). Modal decomposition can be achieved by applying appropriate low-pass filters to remove acceleration content associated with the very small displacement vibrations.

If seat pan data is used to develop a mathematical scale factor, the change in velocities and accelerations of the seat pan associated with the relative displacement oscillations across the spring-damper assembly should not be removed by low-pass filtering. In this report the example seat pan data was filtered using a 30 Hz low-pass filter. This is a conservative approach that assumes that the resulting relative displacements across the spring-damper assembly are relevant to the study of human comfort and lumbar responses in the seat.

Equation (1) computes the complex transfer function H(s)) that can be used to compare the effectiveness of different shock mitigation seats in a wave impact environment. If the complex transfer function is not used, the alternative empirical fit equations that scale from the deck to the seat pan, like those shown in Figures 5 or 9 can be used to evaluate and compare seat mitigation characteristics. The slopes of the data fit equations in Figures 5 and 9 provide a measure of the characteristics of the seat's spring-damper assembly that can be compared directly with other seat data fit equations. Seats with smaller pan-to-deck ratios (i.e., slopes) have better shock response characteristics.

Caution is advised when using any data fitting approach to ensure that the range of applicability of the data fit equation is not exceeded.

The compliance of soft seat cushion material results in relative displacements between the seat pan and the top of the cushion that can cause load amplification in a wave impact environment. The total change in impulse will be the same for cushioned seat or hard seat conditions, but a higher load will be applied for a shorter period of time for a cushioned seat. The selection of seat cushion materials is therefore a compromise between soft-compliant materials that provide comfort and harder seat materials that prevent or limit impact load amplification.

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